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DESIGN CHARTS FOR CROSS-FLOW TUBULAR INTERCOOLERS

CHARGE-ACROSS-TUBE TYPE

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SUMMARY

On the basis of current heat-transfer theory, equations are presented relating the various dimensions, the air mass flow, and the performance of a cross-flow tubular intercooler in which the charge flows across and the cooling air through the tubes. Based on these equations, design charts are presented from which the intercooler design characteristics and the intercooler performance can be quickly determined.

A unit was tested consisting of six staggered banks, each consisting of 28 phosphor-bronze tubes having a length of 8 inches, an outer diameter of 0.313 inch, and a wall thickness of 0.006 inch. The spacing between the tubes was 0.012 inch. The performance obtained with this test unit closely checked the calculated performance and indicated that intercoolers of the annular type having a relatively small number of tube banks may be both practicable and efficient without the addition of excessive weight or volume to aircraft-engine supercharger installations.

The performance of a cross-flow tubular intercooler is shown to be practically independent of the fuel-air ratio of the engine charge at a charge outlet temperature of 140° F.

INTRODUCTION

Among the various devices for increasing the power of the aircraft engine, the intercooler holds a prominent place. Its usefulness is twofold: (1) Cooling the engine charge increases its density and hence the charge energy; and (2), decreasing the charge temperature permits higher manifold pressures without the occurrence of knock. The intercooler demands, however, a certain amount of power expenditure owing to its resistance to the flow of cooling and charge air and to the power required to transport it. In order to have a large net power gain due to the intercooler, the design must be such that the weight and the resistance to air flow are small and the temperature drop of the charge air (or gasoline-air mixture) is large. This design problem involves a number of variables such as the tube length-diameter ratio, the tube spacing, the number and the arrangement of the tubes, the pressure drops through the intercooler, the air mass flow, and the intercooler weight.

No attempt seems to have been made to correlate all these variables with intercooler performance in such a manner as to be of direct assistance to the designer. A considerable amount of information is available concerning the phenomena attending the flow of air through and across metal tubes. This information may be used to predict the performance of conventional tubular intercoolers in terms of the various design factors. It is the purpose of this paper to present a method based on current heat-transfer theory by which such predictions can readily be made. This method is supported by test results obtained from a laboratory intercooler test unit having a comparatively small number of tube banks and closely spaced tubes.

The tests and the analysis were made from October 1939 to June 1940 at the Langley Memorial Aeronautical Laboratory.

APPARATUS AND METHODS

Figure 1 is a photograph of the test assembly used in the investigation. The test equipment consists essentially of a centrifugal blower driven by a 30-horsepower direct-current motor, an intercooler test unit, an electric air heater, a carburetor, and various measuring instruments. The blower is capable of moving 5500 cubic feet of air per minute against a pressure of 40 inches of water. As the diagrammatic sketch in figure 2 shows, the cooling air and the heated charge air (or gasoline-air mixture) were drawn through and across the intercooler tubes, respectively, by this blower, the relative quantities being controlled by valves as shown in the figure. In a large portion of the tests, air alone was drawn across the intercooler tubes. A few runs were made in which predetermined quantities of gasoline were introduced into this air stream to produce various fuel-air ratios. The air or the gasoline-air mixture that was cooled will be referred to as the "charge." The gasoline was introduced into the heated air stream by means of a carburetor, and a Rotameter was used to measure its rate of flow. (See fig. 2.)

The charge air was heated with an electric air heater consisting of five units of 2500 watts each, wired so that any number of them could be employed. The entrance-air and the exit-air temperatures were measured by thermocouples placed at various positions in the cross section of the air or the mixture streams so that good averages could be obtained. (See fig. 3.) The thermocouple connections were so arranged that important temperature differences could be obtained almost instantaneously. The tube-wall temperatures were measured by 34 thermocouples so placed along various tubes that a good average could be obtained. (See fig. 4.) Air mass-flow measurements were made by means of orifice plates in the air streams in accordance with the procedure outlined by the American Society of Mechanical Engineers (reference 1). Pressure data were obtained with water manometers connected to various points in the system, as shown in figure 3. The pressure drop of cooling air in the intercooler tubes was determined by means of a 0.030-inch copper tube, which was prepared for static-pressure measurement and was extended through one intercooler tube, the static pressure being taken at 1/2 inch from either end of the tube. The tube was plugged at A and had static-pressure holes at B and C. (See fig. 3.)

The intercooler test block consisted of 168 phosphor-bronze tubes, which had a length of 8 inches exposed to the charge stream, an outer diameter of 0.313 inch, and a wall thickness of 0.006 inch. (See figs. 3 and 4.) The tubes were arranged in six 28-tube banks, across which the charge flowed. The tubes were supported at the ends by stainless-steel plates of 0.125-inch thickness, the minimum distance between adjacent tubes being 0.012 inch. The banks were staggered so that the tube centers lay on the apexes of equilateral triangles.

Tests were made on the intercooler unit for the purpose of checking experimentally the cooling effectiveness and the pressure drops calculated from the heat-transfer theory. The effect of fuel-air ratio of the charge was also determined. The test conditions covered are given in the following table:

Fuel-air ratio	Rate of cooling-air flow, M_1 (lb/sec)	Rate of charge flow, M_2 (lb/sec)	Charge temperature differential above T_0 at entrance to intercooler, T_2 (°F)
0	0.50	0.08, 0.12, 0.17	300
0	.60	.08, .12, .17	300
0	.70	.08, .12, .17	300
0	.50	.12	156
0	.60	.12	156 to 316
0	.70	.12	156 to 312
.050	.60	.16	261
.057	.60	.16	253
.067	.60	.16	254
.080	.60	.16	246
.100	.60	.16	237
.125	.60	.16	227

SYMBOLS

V_a velocity at entrance of tube, feet per second

V_b velocity before entrance of tube, feet per second

V_c velocity at exit of tube, feet per second

V_d velocity after exit of tube, feet per second

H heat-transfer rate, Btu per second

h_1 heat-transfer coefficient of cooling air, Btu per second per square foot per $^{\circ}F$

h_2 heat-transfer coefficient of charge, Btu per second per square foot per $^{\circ}F$

M_1 rate of cooling-air flow, pounds per second

M_2 rate of charge flow, pounds per second

x distance along a tube, feet

l tube length, feet

d_1 inside tube diameter, foot

d_2 outside tube diameter, feet

d average tube diameter, feet $\left[\frac{1}{2}(d_1 + d_2)\right]$

t tube-wall thickness, feet

s minimum distance between walls of adjacent tubes, feet (see fig. 4(a))

m number of tube banks

n number of tubes in each bank

N number of tubes (mn)

W_t weight of intercooler tubes, pounds

A_1 cross-sectional area of an intercooler tube, square feet $(\pi d^2/4)$

A_2 total area of tube clearance, square feet (nsl)

A_F frontal area of tube block, square feet

f ratio of total cross-sectional area of intercooler tubes to frontal area of intercooler block (NA_1/A_F)

ϵ contractional-loss coefficient

g acceleration of gravity, feet per second per second

c_p specific heat of air at constant pressure, (0.24 Btu per pound per °F)

k thermal conductivity of air, Btu per second per square foot per °F per foot

μ absolute viscosity of air, pounds per second-foot

Δp_1 total pressure drop of cooling air through the intercooler, inches of water

Δp_2 total pressure drop of charge across the tube banks of the intercooler, inches of water

Δp_{en} pressure drop of cooling air at tube entrance, inches of water

Δp_{fr} pressure drop of cooling air due to friction in tubes, inches of water

Δp_H pressure drop of cooling air due to heating in tubes, inches of water

Δp_{ex} pressure drop of cooling air at tube exit, inches of water

P_1 power required to force cooling air through intercooler tubes, horsepower

P_2 power required to force charge across intercooler-tube banks, horsepower

T_o cooling-air temperature at tube entrance, °F

T_1 cooling-air temperature differential above T_o at tube exit, ^oF

T_w mean tube-wall temperature differential above T_o, ^oF

T_2 charge temperature differential above T_o at entrance to intercooler, ^oF

$T_{2_{ex}}$ charge temperature differential above T_o at exit of intercooler, ^oF

η cooling effectiveness, $\dfrac{T_2 - T_{2_{ex}}}{T_2}$

ρ_o standard atmospheric density at 59° F and 29.92 inches of mercury (0.0765 pound per cubic foot)

$\rho_{1_{en}}$ cooling-air density at entrance of tubes, pound per cubic foot

$\rho_{1_{ex}}$ cooling-air density at exit of tubes, pounds per cubic foot

$\rho_{1_{av}}$ arithmetic mean of $\rho_{1_{en}}$ and $\rho_{1_{ex}}$, pounds per cubic foot

$\rho_{2_{av}}$ mean charge density in intercooler, pounds per cubic foot

σ_1 density of entrance cooling air relative to standard atmoshpere $(\rho_{1_{en}}/\rho_o)$

σ_2 mean density of charge in intercooler relative to standard atmosphere $(\rho_{2_{av}}/\rho_o)$

Unless otherwise defined, the subscripts 1 and 2 refer to cooling air and charge, respectively.

ANALYSIS

Cooling Effectiveness

At some distance x along the length of an intercooler tube the amount of heat per second dH exchanged between the charge and the cooling air through an elemental length of tube dx is:

$$dH = h_2 \pi d_2 \, dx \, (T_{2_{av}} - T_w) = h_1 \pi d_1 \, dx \, (T_w - T_x) =$$

$$= A_1 \rho_1 V_1 c_p \, dT_x \qquad (1)$$

where $T_{2_{av}}$ is the mean temperature above T_c of charge across the tube and T_x is the temperature above T_o of the cooling air at distance x along the tube.

Also, the total heat given up by the charge to the first bank of tubes is

$$H = n \, s \, l \, \rho_2 V_2 c_p (T_2 - T_2') = n A_1 \rho_1 V_1 c_p T_1 \qquad (2)$$

where T_2' is the temperature of the charge after the first bank and T_1 is the temperature of cooling air at $x = l$.

When T_w is eliminated in equation (1)

$$\frac{dT_x}{T_x - T_{2_{av}}} = \frac{-\pi h_1 d_1 h_2 d_2 \, dx}{A_1 c_p \rho_1 V_1 (h_1 d_1 + h_2 d_2)} \qquad (3)$$

Let

$$\frac{\pi h_1 d_1 h_2 d_2}{A_1 c_p \rho_1 V_1 (h_1 d_1 + h_2 d_2)} = c \qquad (4)$$

Then, when equation (3) is integrated,

$$T_x = T_{2_{av}} (1 - e^{-cx}) \qquad (5)$$

Because $T_{2_{av}} = \frac{T_2 + T_2'}{2}$, equation (5) becomes

$$T_x = \frac{1}{2} (T_2 + T_2') (1 - e^{-cx}) \tag{6}$$

When equation (6) is substituted in equation (2) and when the total length of the tube is considered

$$s\, l\, \rho_2\, V_2\, c_p\, (T_2 - T_2') =$$

$$= \frac{1}{2} A_1\, \rho_1\, V_1\, c_p\, (T_2 + T_2') (1 - e^{-cl}) \tag{7}$$

The solution of equation (7) for T_2' is

$$T_2' = T_2 \left[\frac{1 - \frac{r}{2} (1 - e^{-cl})}{1 + \frac{r}{2} (1 - e^{-cl})} \right] \tag{8}$$

where

$$r = \frac{A_1\, \rho_1\, V_1}{s\, l\, \rho_2\, V_2} = \frac{M_1}{M_2 m}$$

Likewise, after the second bank,

$$T_2'' = T_2' \left[\frac{1 - \frac{r}{2} (1 - e^{-cl})}{1 + \frac{r}{2} (1 - e^{-cl})} \right]$$

or, from equation (8),

$$T_2'' = T_2 \left[\frac{1 - \frac{r}{2} (1 - e^{-cl})}{1 + \frac{r}{2} (1 - e^{-cl})} \right]^2$$

and so on, until after the m^{th} bank, where

$$T_{2_{ex}} = T_2 \left[\frac{1 - \frac{r}{2} (1 - e^{-cl})}{1 + \frac{r}{2} (1 - e^{-cl})} \right]^m$$

Therefore, the cooling effectiveness is

$$\eta = \frac{T_2 - T_{2_{ex}}}{T_2} = 1 - \left[\frac{1 - \frac{r}{2}(1 - e^{-cl})}{1 + \frac{r}{2}(1 - e^{-cl})}\right]^m$$

or

$$\eta = 1 - \left[\frac{2}{1 + \frac{M_1}{2m\,M_2}(1 - e^{-cl})} - 1\right]^m \tag{9}$$

From equation (9), it is seen that the cooling effectiveness is determined by the mass flow of the cooling air and the charge, the number of tube banks, the tube length, and the quantity c, which, from equation (4), may be expressed:

$$c = \frac{\pi\, h_1\, d_1\, h_2\, d_2\, N}{c_p\, M_1\, (h_1\, d_1 + h_2\, d_2)}$$

because

$$A_1\, \rho_1\, V_1 = \frac{M_1}{N}$$

For thin-wall tubing, such as that used in intercoolers, the average tube diameter may be substituted for the inside and the outside tube diameters, that is, $d_1 = d_2 = d$. The exponent cl in equation (9) may then be expressed as follows:

$$cl = \frac{\pi\, d\, h_1\, h_2\, N\, l}{(h_1 + h_2)\, c_p\, M_1} = \frac{\dfrac{\pi\, d\, h_1\, N\, l}{M_1\, c_p}}{1 + \dfrac{h_1}{h_2}} \tag{10}$$

The heat-transfer coefficients h_1 and h_2 may be evaluated in terms of the intercooler dimensions, the air mass flow, and certain physical properties of the charge and the cooling air by referring to the research summarized in reference 2 (p. 169) in which

$$\frac{h_1 d}{k_1} = 0.0225\left(\frac{\rho_1\, V_1 d}{\mu_1}\right)^{0.8}\left(\frac{c_p\, \mu_1}{k_1}\right)^{0.4} \tag{11}$$

Because $c_p\mu_1/k_1$ for air is practically constant at 0.73, equation (11) may be written

$$h_1 = 0.01984 \frac{k_1}{\mu_1^{0.8}} \frac{(\rho_1 V_1)^{0.8}}{d^{0.2}} \qquad (12)$$

The factor $k_1/\mu_1^{0.8}$ shows little variation over the range of cooling temperatures that is usually encountered in intercoolers and a constant value of $k_1/\mu_1^{0.8} = 0.0323$ at 59° F will be used.

Because $\rho_1 V_1 = \frac{4M_1}{\pi d^2 N}$ equation (12) becomes

$$h_1 = 0.777 \times 10^{-3} \frac{M_1^{0.8}}{N^{0.8} d^{1.8}} \qquad (13)$$

If for the present the effect of the presence of fuel vapor is neglected, the heat-transfer coefficient between the charge and the tubes may be obtained from the following expression given in reference 2 (p. 227):

$$\frac{h_2 d}{k_{2_f}} = 0.131 \left(\frac{\rho_2 V_2 d}{\mu_{2_f}}\right)^{0.69} \qquad (14)$$

The subscript f indicates that the physical properties of the charge air are based on a film temperature, which is an average of the mean charge air and the tube-wall temperatures.

When the factor $\frac{k_{2_f}}{\mu_{2_f}^{0.69}}$ is taken at a film temperature of 100° F at 0.00951 and because $\rho_2 V_2 = \frac{M_2 m}{Nsl}$ equation (14) becomes

$$h_2 = \frac{0.1246 \times 10^{-2}}{d} \left(\frac{M_2 md}{Nsl}\right)^{0.69} \qquad (15)$$

An investigation of the effect of the air temperature on the cooling effectiveness showed that, when the thermal conductivity k and the absolute viscosity μ in equation (11) are evaluated at 100° F instead of 59° F and in equation (14) at 200° F instead of 100° F, the difference between the effectivenesses is less than 1 percent, which is well within experimental accuracy.

When the values of h_1 and h_2, as given by equations (13) and (15), respectively, are substituted in equation (10), there results

$$cl = \frac{1.017 \times 10^{-2} \left(\frac{Nd}{M_1}\right)^{0.2} \left(\frac{l}{d}\right)}{1 + 0.624 \left(\frac{M_1}{M_2}\right)^{0.69} \left(\frac{M_1}{Nd}\right)^{0.11} \left(\frac{s}{md}\right)^{0.69} \left(\frac{l}{d}\right)^{0.69}} \tag{16}$$

The application of the binomial theorem to equation (9) shows that the variation in m as it appears explicitly in equation (9) has a negligible effect on the cooling effectiveness. The influence of m on the cooling effectiveness is through its effect on the exponent cl. The factors upon which the cooling effectiveness depends, therefore, are shown in equations (9) and (16) to be Nd/M_1, l/d, M_1/M_2, and md/s.

In order to make graphical representations less complicated, the variable Nd/M_1, which has a relatively small effect on the cooling effectiveness, may be accounted for by letting

$$\left(\frac{l}{d}\right)\left(\frac{Nd}{M_1}\right)^{0.2} = 10^{0.2} \left(\frac{l}{d}\right)_{eq}$$

$$\left(\frac{md}{s}\right)\left(\frac{Nd}{M_1}\right)^{0.36} = 10^{0.36}\left(\frac{md}{s}\right)_{eq}$$

where the subscript eq means equivalent.

It is evident from the preceding equations that, when $Nd/M_1 = 10$, $\left(\frac{l}{d}\right)_{eq} = \left(\frac{l}{d}\right)$ and $\left(\frac{md}{s}\right)_{eq} = \left(\frac{md}{s}\right)$. For values of Nd/M_1 other than 10, plots of the preceding equations give the corrections to be applied to l/d and md/s to obtain $\left(\frac{l}{d}\right)_{eq}$ and $\left(\frac{md}{s}\right)_{eq}$.

Equation (16) can now be written as:

$$cl = \frac{0.01612 \left(\frac{l}{d}\right)_{eq}}{1 + 0.484 \left(\frac{M_1}{M_2}\right)^{0.69} \left(\frac{md}{s}\right)_{eq}^{-0.69} \left(\frac{l}{d}\right)_{eq}^{0.69}} \tag{17}$$

Thus equation (9) may be written

$$\eta = \Phi_o \left[\left(\frac{l}{d}\right)_{eq}, \left(\frac{md}{s}\right)_{eq}, \frac{M_1}{M_2} \right]$$

Charts based on equations (9) and (17) will be presented to show graphically the effect of the variables involved on cooling effectiveness. The experimental verification of equation (9) will be discussed later in this report.

Pressure Drop of Cooling Air

Entrance losses. - One cause of the pressure drop at the tube entrance is the conversion of the static head into dynamic head because of the reduction of the flow area. From Bernoulli's equation, this drop in pressure is

$$\Delta p_{en}' = \frac{\rho_{1_{en}}}{5.2 \times 2g} (V_a^2 - V_b^2)$$

Because

$$V_b = \frac{M_1}{\rho_{1_{en}} A_F}, \quad V_a = \frac{M_1}{\rho_{1_{en}} f A_F}, \text{ and } \quad fA_F = \frac{\pi d^2 N}{4}$$

$$\frac{\rho_{1_{en}}}{\rho_o} \Delta p_{en}' = \frac{1}{10.4 \ \rho_o g} \left(\frac{4M_1}{\pi d^2 N}\right)^2 (1 - f^2) \tag{18}$$

There is also a drop in the pressure near the entrance attributed to the increase in maximum velocity that results from the change in velocity distribution along the tube diameter. This drop in pressure, according to reference 3 (p. 51), is

$\Delta p_{en}'' = \frac{0.09 \ \rho_{1_{en}}}{10.4 \ g} V_a^2$, which may be written as:

$$\frac{\rho_{1_{en}}}{\rho_o} \Delta p_{en}'' = \frac{0.09}{10.4 \ \rho_o g} \left(\frac{4M_1}{\pi d^2 N}\right)^2 \tag{19}$$

The contraction of the air stream immediately after it enters the tube causes a loss, which further reduces the pressure by

$$\Delta p_{en}''' = \frac{\epsilon \rho_{1_{en}}}{10.4\ g} V_a^2$$

where ϵ is a function of the area ratio f. Values of ϵ were taken from reference 2 (pp. 121-122). This equation may be written by eliminating V_a

$$\frac{\rho_{1_{en}}}{\rho_o} \Delta p_{en}''' = \frac{\epsilon}{10.4\ \rho_o g} \left(\frac{4M_1}{\pi d^2 N}\right)^2 \qquad (20)$$

Frictional loss in the tubes. - The pressure drop due to fluid friction Δp_{fr} is given by the Fanning equation as

$$\Delta p_{fr} = \frac{6.49\ M_1^2\ l\ F}{10.4\ g\ \rho_{1_{av}}\ N^2\ d^5}$$

where (from reference 2, p. 111) the friction factor is

$$F = 0.049\ (R)^{-0.2} \quad (5000 < R < 200,000)$$

Because the Reynolds number

$$R = \frac{\rho_{1_{av}} V_1 d}{\mu_{1_{av}}} = \frac{4M_1}{\pi d N\ \mu_{1_{av}}}$$

and $\mu_{1_{av}} = 120 \times 10^{-7}$ at 59° F,

$$\frac{\rho_{1_{av}}}{\rho_o} \Delta p_{fr} = \frac{0.01946}{10.4\ \rho_o g} \left(\frac{4M_1}{\pi d^2 N}\right)^2 \left(\frac{l}{d}\right) \left(\frac{Nd^2}{M_1}\right)^{0.2} \left(\frac{1}{d}\right)^{0.2} \qquad (21)$$

If $\dfrac{\rho_{1_{av}}}{\rho_o} = \dfrac{(2 + \beta)}{(2 + 2\beta)} \dfrac{\rho_{1_{en}}}{\rho_o}$ is substituted in equation (21) where $\beta = \dfrac{T_1}{T_o + 460}$

$$\frac{\rho_{1_{en}}}{\rho_o} \Delta p_{fr} = \frac{0.01946}{10.4\rho_o g} \left(\frac{2+2\beta}{2+\beta}\right) \left(\frac{4M_1}{\pi d^2 N}\right)^2 \left(\frac{l}{d}\right)\left(\frac{Nd^2}{M_1}\right)^{0.2} \left(\frac{1}{d}\right)^{0.2} \tag{22}$$

If, in the calculation of the frictional loss in equation (22), the absolute viscosity were evaluated at 100° F instead of at 59° F, the frictional-loss increase would be only 1.2 percent, which is well within experimental accuracy.

A slight reduction in pressure in the tubes occurs because of the velocity increase resulting from the progressive increase in cooling-air temperature when the cooling air traverses the length of the tube. This pressure decrease (by equating force to the rate of momentum change) is

$$\Delta p_H = \frac{\rho_{1_{en}} V_a}{5.2\ g} (V_c - V_a) = \frac{\rho_{1_{en}} V_a^2}{5.2\ g} \left(\frac{V_c}{V_a} - 1\right)$$

Because $V_c = \dfrac{\rho_{1_{en}}}{\rho_{1_{ex}}} V_a$

$$V_c = \left(\frac{T_o + T_1 + 460}{T_o + 460}\right) V_a$$

or

$$V_c = \left(1 + \frac{T_1}{T_o + 460}\right) V_a = (1 + \beta)\ V_a$$

Then

$$\frac{\rho_{1_{en}}}{\rho_o} \Delta p_H = \frac{\beta}{5.2\ \rho_o g} \left(\frac{4M_1}{\pi d^2 N}\right)^2 \tag{23}$$

Exit losses. - The pressure drop at the tube exit may be derived by equating force to the rate of change in momentum

$$\Delta p_{ex} = \frac{\rho_{1_{ex}} V_d}{5.2\ g} (V_d - V_o)$$

If

$$V_d = \frac{M_1}{\rho_{1_{ex}} A_F},\ V_c = \frac{M_1}{\rho_{1_{ex}} f A_F},\ \text{and}\ \ \rho_{1_{ex}} = \frac{\rho_{1_{en}}}{1 + \beta}$$

are substitutod in the foregoing equation

$$\frac{\rho_{1_{en}}}{\rho_o} \Delta p_{ex} = \frac{(1 + \beta)}{5.2\ \rho_o g} \left(\frac{4M_1}{\pi d^2 N}\right)^2 (f^2 - f) \tag{24}$$

which is evidently a negative pressure drop, that is, a pressure rise.

Pressure Drop of Charge

The pressure drop of charge across tube banks in staggered rows, according to reference 2 (p. 120), is

$$\Delta p_2 = \frac{4\rho_{2_{av}} \left(V_{2_{max}}\right)^2 m f'''}{5.2\ (2g)}$$

where $V_{2_{max}}$ is the velocity of charge through the minimum space between the tubes and

$$f''' = 0.8 \left(\frac{s\rho_{2_{av}} V_{2_{max}}}{\mu_{2_{av}}}\right)^{-0.22}$$

Because

$$V_{2_{max}} = \frac{M_2 m}{l\ N\ s\ \rho_{2_{av}}}$$

and

$$\mu_{2_{av}} = 127 \times 10^{-7} \text{ at } 100^{\circ}\ F$$

it follows that

$$\frac{\rho_{2_{av}}}{\rho_o} \Delta p_2 = \frac{0.1344m^{0.78}}{5.2\ \rho_o\ g} \left(\frac{M_2}{Nd l}\right)^{1.78} \left(\frac{md}{s}\right)^2 \left(\frac{1}{d}\right)^{0.22} \tag{25}$$

If, in the calculation of the frictional loss in equation (25), the absolute viscosity were evaluated at 200° F instead of at 100° F, the frictional-loss increase would be only 2.8 percent, which is well within experimental accuracy.

Intercooler Design Charts

An examination shows that the equations for effectiveness and pressure drops may be expressed as

$$\eta = \Phi\left(\frac{l}{d}, \frac{M_1}{M_2}, \frac{md}{s}, \frac{Nd^2}{M_1}, \frac{1}{d}, m\right) \quad \text{(see equations (9) and (16))} \tag{26}$$

$$\sigma_1 \Delta p_1 = \Phi'\left(\frac{M_1}{Nd^2}, \frac{l}{d}, f, \beta, \frac{1}{d}, \epsilon\right) \quad \text{(see equations (13) through (24))} \tag{27}$$

$$\sigma_2 \Delta p_2 = \Phi''\left(\frac{M_2}{Nd l}, \frac{md}{s}, m, \frac{1}{d}\right) \quad \text{(see equation (25))} \tag{28}$$

The weight of intercooler tubes W_t is $\pi\ d\ l\ N\ t\ \rho_t$ where ρ_t is the tube-wall density, pounds per cubic foot. Then

$$\frac{W_t}{\pi\ t\ \rho_t} = \alpha = Ndl = \text{weight factor}$$

and

$$Nd^2 = \frac{\alpha\ d}{l}$$

By definition,

$$f = \frac{\text{total tube cross-sectional area}}{\text{frontal area of intercooler block}}$$

or

$$f = \frac{\pi\, d_1^{\,2}\, N}{4\,(d_2 + s)\,\frac{N}{m}\left[0.866\,(d_2 + s)\,(m - 1) + d_2\right]}$$

$$f = \frac{\pi\, m}{4\left(1 + \frac{\frac{s}{d} + 2\frac{t}{d}}{1 - \frac{t}{d}}\right)\left[0.866\left(1 + \frac{\frac{s}{d} + \frac{2t}{d}}{1 - \frac{t}{d}}\right)(m-1) + \left(1 + \frac{2\frac{t}{d}}{1 - \frac{t}{d}}\right)\right]} \qquad (29)$$

A plot of equation (29), which will be discussed later, shows that any variation in t/d over the range used in intercooler practice has little effect on f, the area ratio. Accordingly, t/d was taken at 0.02 in equation (29), so that

$$f = \psi\left(\frac{s}{d}, m\right) \qquad (30)$$

It was previously stated that ϵ is a function of f and that the expansion of equation (9) into a series by means of the binomial theorem shows that the effect of m in equation (9) is negligible. In expressions (26), (27), and (28) that l/d terms have small exponents, so that these terms, for any variation in tube diameter within the range used in aircraft intercoolers, have little effect on the effectiveness and the pressure drops. The principal effect of the variation of tube diameter is obtained through its influence on the l/d and the md/s terms. For this reason the tube diameter for the l/d term was taken at an average value of 0.30 inch. The factor β is involved in the pressure loss in the intercooler tubes due to the heating of the cooling air, which is a small portion of the total loss. This factor was evaluated as 0.06 in this analysis and should represent an average condition.

When expressions (26), (27), and (28) are rewritten on the basis of the preceding discussion,

$$\eta = \psi'\left(\frac{l}{d}, \frac{M_1}{M_2}, \frac{md}{s}, \frac{M_2}{\alpha}\right) \qquad (31)$$

$$\sigma_1\, \Delta p_1 = \psi''\left(\frac{M_2}{\alpha}, \frac{l}{d}, f, \frac{M_1}{M_2}\right) \qquad (32)$$

$$\sigma_2 \, \Delta p_2 = \psi''' \left(\frac{M_2}{\alpha}, \ \frac{md}{s}, \ m\right) \qquad (33)$$

The terms in expressions (31), (32), and (33) are the basic variables considered in the construction of the design charts. Inspection of expression (33) shows that, for an assumed value of $\sigma_2 \, \Delta p_2$ and m, the value of md/s and hence f (expression (30)) is determined by M_2/α. From expression (32) it may be seen that for a given value of M_2/α, f, and $\sigma_1 \, \Delta p_1$ the value of l/d determines M_1/M_2. Thus all the terms upon which η depends (expression (31)) have been evaluated.

The power required to force cooling air through the intercooler is

$$P_1 = \frac{5.2 \, M_1 \, \Delta p_1}{550 \, \rho_{1_{av}}}$$

or

$$\frac{\rho_{1_{av}}}{\rho_o} \frac{P_1}{M_2} = \left(\frac{M_1}{M_2}\right) \frac{5.2 \, \sigma_1 \, \Delta p_1}{550 \, \rho_{1_{av}}}$$

But

$$\frac{\rho_{1_{av}}}{\rho_o} = \sigma_1 \left(\frac{2 + \beta}{2 + 2\beta}\right)$$

then

$$\frac{\sigma_1^{\,2} \, P_1}{M_2} = \left(\frac{2 + 2\beta}{2 + \beta}\right) \frac{M_1}{M_2} \, \frac{5.2 \, \sigma_1 \, \Delta p_1}{550 \, \rho_o} \qquad (34)$$

or, when the power is expressed as a percentage of the engine brake horsepower for a fuel-air ratio of 0.08 and a specific fuel consumption of 0.5 pound per brake horsepower per hour.

$$\frac{\sigma_1^{\,2} \, P_1 \times 100}{\text{brake horsepower}} = \frac{100}{576} \left(\frac{\sigma_1^{\,2} \, P_1}{M_2}\right) \qquad (35)$$

Likewise, the power required to force the charge across the staggered-tube banks of the intercooler may be expressed as:

$$\frac{\sigma_2{}^2 \, P_2}{M_2} = \frac{5.2 \, \sigma_2 \, \Delta p_2}{550 \, \rho_0}$$

and for a fuel-air ratio of 0.03 and a specific fuel consumption of 0.50 pound per brake horsepower hour as:

$$\frac{\sigma_2{}^2 \, P_2 \, (100)}{\text{brake horsepower}} = \frac{100}{576} \left(\frac{\sigma_2{}^2 \, P_2}{M_2}\right) \tag{36}$$

The dimensions of this type of intercooler may be expressed as follows:

The dimension of the block in the direction of charge flow is

$$l_e = 0.866 \, (d_2 + s) \, (m - 1) + d_2 \tag{37}$$

The width of the block is

$$w = \frac{N}{m} \, (d_2 + s) \tag{38}$$

The volume of the block therefore is

$$v = l \, l_e \, w \tag{39}$$

because l is the dimension in the direction of cooling-air flow.

RESULTS AND DISCUSSION

Test Results

Figure 5(a) shows the effect of mass flow of cooling air and charge on the cooling effectiveness of the intercooler test unit shown in figures 3 and 4. The curves were computed from equation (9); the designated points represent the experimental values. Tests were made for various values of charge inlet temperature, but no differences in cooling effectiveness were obtained beyond the limits of experimental error. The results of the variation of cooling effectiveness with charge inlet temperature are shown in figure 5(b). Figure 5(c) shows the results obtained by flowing gasoline-air mixtures across the intercooler unit instead of air alone. The small effect of fuel-air ratio on cooling effectiveness may be noted. The temperature of the charge leaving the intercooler was approximately 140° F in the tests in which the fuel-air ratio was varied.

The pressure drop sustained by a given mass flow of gasoline-air mixture across the intercooler unit was slightly less than the pressure drop for an equal mass flow of air. This phenomenon is an indication that fuel did not collect on the intercooler tubes because an increased pressure drop would be expected if the tube spaces were clogged with condensed gasoline. The absence of an increased pressure drop cannot, however, be taken as a definite indication of the lack of condensation because of the possibility that the high charge velocity may prevent any accumulation of gasoline on the tube walls. Inspection of the tube block after runs showed no evidence of liquid gasoline about the intercooler, and it is believed that no condensation occurred. At low charge temperatures appreciable condensation might occur.

Figure 6 shows the pressure drop of charge air across the intercooler test unit as a function of the rate of charge flow. The computed curve is based on equation (25) and the experimental curve is drawn through the test points.

Figure 7 shows the several pressure changes in the cooling air when it passes through the intercooler unit as a whole. The solid lines join the values computed by means of equations (18), (19), (20), (22), (23), and (24); the indicated points denote test values. As a whole, good agreement between the computed and the experimental data is shown.

Figure 8, which is based on equations (9) and (17), shows that η increases with M_1/M_2, $(l/d)_{eq}$, and $(md/s)_{eq}$ at a rate that diminishes as these variables increase. The effect of l/d and md/s on η is nearly the same as that of $(l/d)_{eq}$ and $(md/s)_{eq}$. The corrections to be applied to l/d and md/s to obtain $(l/d)_{eq}$ and $(md/s)_{eq}$ are plotted in figure 9 as functions of Nd/M_1. Equations presented earlier in this report indicate that an increase in either l/d or md/s is accompanied by an increase in pressure drop, either through or across the intercooler. Therefore, an attempt to attain higher cooling effectivenesses by increasing l/d or md/s is curbed by reasonable limits of pressure drop.

The equations relating the pressure drop of cooling air and charge to various design factors are plotted in figures 10 and 11, respectively. Figure 12 shows the variation of the area ratio f with s/d, t/d, and m (from equation (29)). The contractional-loss coefficient ϵ used in calculating a part of the cooling-air entrance losses is replotted in figure 13 (from reference 2, pp. 121-122).

Intercooler Design

Figures 8 to 12 are cumbersome for the intercooler designer to use. One purpose of this report is, therefore, to present the design information in such form (figs. 14, 15, and 16) as to reduce the complexities of design to a minimum. In order to make these simplifications, it has been necessary to make certain approximations that, to a designer, should be inappreciable, since the approximations will not introduce errors outside the region of experimental accuracy. These approximations have been discussed in the section of the analysis devoted to the development of the equations upon which figures 14, 15, and 16 are based. Figure 14 is based on the arrangement of five banks of tubes. The excessive block width necessitated by such a small number of banks is suitable only for the annular type of intercooler installation, which will be discussed later. Figure 15, based on 30 tube banks, may be used for more compact types of intercooler block. Cooling effectiveness for a number of banks between 5 and 30 may be obtained by linear interpolation between figures 14 and 15 without appreciable error. Figures 14 and 15 show the cooling effectiveness plotted against the factor Ndl/M_2, which is an index of the heat-transfer surface and also of the tube weight for any particular engine power. Each plot is made for constant values of $\sigma_1 \Delta p_1$ and m, but the values of l/d and M_1/M_2 vary. All the plots are made for a constant value of $\sigma_2 \Delta p_2$ (10 in. of water), but the cooling effectiveness for other values of $\sigma_2 \Delta p_2$ may be found by applying corrections that are given in figure 16(a), in which the cooling-effectiveness correction is plotted against $\sigma_2 \Delta p_2$ for different values of l/d and $\sigma_1 \Delta p_1$. The method of making such corrections will be illustrated later in this report. In figures 14 and 15 the increase in cooling effectiveness with cooling surface is to be expected. Furthermore, with constant tube surface the cooling decreases as l/d becomes larger when the pressure drops through and across the intercooler are constant. If M_1/M_2, as well as the pressure drops, is constant, however, the cooling effectiveness increases with l/d but only at the cost of increased cooling area and thus at the cost of increased tube weight.

The tables at the top of figures 14 and 15 show the power expenditures involved in the operation of an intercooler. The maximum drag involved in forcing cooling air through the tubes is shown to be a little over 1 percent of the engine brake horsepower. These tables and figure 16(b) also indicate that the power required of the supercharger to force the charge across the tubes is less than 0.5 percent of the engine brake horsepower. From this fact it appears that the power expenditure of a properly designed intercooler, aside from that required for transportation, may be secondary in importance to weight and size in many cases.

The abscissa scale on the design charts (figs. 14 and 15) has been converted for a special case into another scale, namely, the intercooler-tube weight, pounds per engine brake horsepower. This conversion was made for an engine fuel-air ratio of 0.080, a specific fuel consumption of 0.5 pound per brake horsepower-hour, and for steel tubes with a wall thickness of 0.006 inch and a density of 0.284 pound per cubic inch by multiplying Ndl/M_2 by

$$[\pi\ (12.5)\ (0.006)\ (0.5)\ (0.284)]\left[\frac{1728}{(3600)\ (12)}\right]$$

the expression $\left[\frac{1728}{(3600)\ (12)}\right]$ being a unit conversion factor. All values occurring in the tables at the top of figures 14 and 15 where brake horsepower is involved are likewise based on a fuel-air ratio of 0.080 and a specific fuel consumption of 0.5 pound per brake horsepower-hour.

The power for forcing the charge through the intercooler is probably of greater importance than that required to force the cooling air because of the reduction in maximum engine power with decreased intercooler discharge pressure. This reduction in maximum engine power may be estimated for a given temperature drop of the charge by assuming the engine power to vary directly with the manifold pressure. The relative importance of these power losses and of those due to intercooler weight and size depends on the design and the speed of the airplane. It was therefore considered advisable to make this information readily available to the designer (figs. 14, 15, and 16) rather than to present optimum designs or a method of obtaining optimum designs based on some artificial figure of merit. An additional factor of importance is the external drag of the cooling-air inlet. This drag can vary from almost zero to several times the cooling-air drag, depending on the inlet locations.

Illustrations of the Use of the Intercooler Design Charts

A procedure that may be followed in using the intercooler design charts is shown by means of several examples. Let it be supposed that an intercooler is to be designed for the following set of conditions.

Case I (sea level)

Engine characteristics:

(1) Brake horsepower 1000

(2) Specific fuel consumption, pounds per brake horsepower per hour 0.50

(3) Fuel-air ratio 0.080

Desired intercooler dimensions:

(4) Length of tube l, feet 5/6

(5) Average diameter of tube d, feet 1/48

(6) Number of tube banks m 5

(7) Tube-wall thickness t, feet (steel density, 490 lb/cu ft) 0.0005

Intercooler limitations:

(8) Cooling-air pressure drop Δp_1, inches water 6.0

(9) Charge pressure drop Δp_2, inches water 10.0

(10) Relative density of cooling air σ_1 . . . . 1.0

(11) Relative density of charge σ_2 . . . . 1.0

Desired intercooler performance:

(12) Cooling effectiveness η, percent 75

It is desired to find the following characteristics of the intercooler:

Number of tubes N

Tube spacing s

Weight of intercooler tubes W_t and dimensions of intercooler block

Power required to force cooling air through the intercooler P_1

Power required to force charge across the tube banks of the intercooler P_2

Weight of cooling air handled by the intercooler M_1

From items (6) and (8) to (11) it is seen that $m = 5$, $\sigma_1 \Delta p_1 = 6$ inches of water, and $\sigma_2 \Delta p_2 = 10$ inches of water. Figure 14 (c) applies for this case.

(13) From items (1), (2), and (3)

$$M_2 = \frac{1000 \times 0.5}{0.08 \times 3600} = 1.736 \text{ pounds per second}$$

(14) From items (4) and (5)

$$\frac{l}{d} = \frac{5/6}{1/48} = 40$$

(15) If figure 14(c) and items (12) and (14) are used

$$\frac{Ndl}{M_2} = 22.5$$

and

(16)
$$\frac{M_1}{M_2} = 4.6$$

from which the cooling-air weight M_1 is 8.0 pounds per second.

(17) If figure 11(a) and item (15) are used

$$\frac{md}{s} = 176$$

(18) Number of tubes

$$N \text{ is } \frac{\text{item (15)} \times M_2}{ld} = \frac{(22.5)\ (1.736)}{(5/6)\ (1/48)} = 2250$$

(19) Tube spacing

$$s = \frac{md}{\text{item (17)}} = \frac{(5)\ \frac{1}{48}}{176} = 0.000592 \text{ foot} = 0.0071 \text{ inch}$$

(20a) Weight of intercooler tubes is item (7) × item (13) × item (15) × $\pi\rho_t$ = (0.0005) (1.736) (22.5) (0.284) (1728π) = 30.1 pounds

(20b) Dimension of intercooler block in direction of charge flow from equation (37) and items (5), (6), (7), and (19) is 0.866 (0.0213 + 0.000592) (5 - 1) + 0.0213 = 0.0972 foot or 1.17 inches

(20c) Width of intercooler block from equation (38) and items (5), (6), (7), (18), and (19) is

$$\frac{2250}{5}\ (0.0213 + 0.000592) = 9.86 \text{ feet}$$

(20d) Volume of intercooler block from equation (39) and item (4) is therefore

$\frac{5}{6} \times 0.0972 \times 9.86 = 0.800$ cubic foot

or 1382 cubic inches

(This arrangement is suitable for intercoolers of the annular type. The diameter of the annulus would be approximately 9.86/3,142 = 3.1 ft.)

(21) From tables at top of figure 14(c) and item (16)

$\frac{\sigma_1{}^2 P_1}{M_2} = 3.51$ horsepower per pound per second of charge flow

(22) Also, $\frac{\sigma_2{}^2 P_2}{M_2} = 1.24$ horsepower per pound per second of charge flow

(23) Cooling power

$$P_1 = \frac{\text{item (21)} \times M_2}{\sigma_1{}^2} = 6.10 \text{ horsepower}$$

(24) Power to force the charge through the intercooler

$$P_2 = \frac{\text{item (22)} \times M_2}{\sigma_2{}^2} = 2.14 \text{ horsepower}$$

(For convenience fig. 14(c) also gives items (20), (23), and (24) as percentages of the brake horsepower based on a specific fuel consumption of 0.50 pound per brake horsepower-hour and a fuel-air ratio of 0.080.)

Case II (13,000 ft)

The design of an intercooler for an altitude of 13,000 feet for the same engine conditions, cooling effectiveness, tube dimensions, and limitations as in case I.

From a table of standard altitudes $\sigma_1 = 0.671$, so that the value of $\sigma_1 \Delta p_1$ is now 0.671×6, which equals 4.0 inches of water. Figure 14(b) is therefore used. Items (13) and (14) of case I also apply here.

(25) If figure 14(b) and items (12) and (14) are used

$$\frac{Nd l}{M_2} = 26.5$$

and

(26) $\dfrac{M_1}{M_2} = 4.3$

from which the weight flow of the cooling air

$$M_1 = 7.5 \text{ pounds per second}$$

(27) If figure 11(a) and item (25) are used

$$\frac{md}{s} = 202$$

(28) Number of tubes

$$N = \frac{\text{item (25)} \times M_2}{l l} = \frac{(26.5)\ (1.736)}{(5)\ (1/48)} = 2650$$

(29) Tube spacing

$$s = \frac{md}{\text{item (27)}} = 0.000514 \text{ foot} = 0.0062 \text{ inch}$$

(30) Weight of intercooler tubes is:

$$\text{item (7)} \times \text{item (13)} \times \text{item (25)} \times \pi\rho_t =$$
$$(0.0005)\ (1.736)\ (26.5)\ (0.284)\ (1728\pi) =$$
35.5 pounds.

(31) From tables at top of figure 14(b) and item (28)

$$\frac{\sigma_1{}^2 P_1}{M_2} = 2.18 \text{ horsepower per pound per second of}$$
charge flow

(32) Also,

$$\frac{\sigma_2^2 P_2}{M_2} = 1.24 \text{ horsepower per pound per second of}$$

charge flow

(33) $P_1 = \dfrac{\text{item (31)} \times M_2}{\sigma_1^2} = 8.42$ horsepower

(34) $P_2 = \dfrac{\text{item (32)} \times M_2}{\sigma_2^2} = 2.14$ horsepower

(If the performance of an intercooler having 30 banks of tubes is desired, fig. 15 is used. For other numbers of banks, linear interpolation between figs. 14 and 15 is sufficiently accurate. The value of md/s for a number of banks other than five is found by multiplying the value in fig. 11(a) by the ratio given in fig. 11(b).)

Case III (sea level)

The effect on intercooler dimensions and performance of a change in pressure drop of the charge $(\sigma_2 \Delta p_2)$ from 10 inches of water (for which figs. 14 and 15 have been drawn) to another value is illustrated by the following case.

Suppose that in case I the charge pressure drop Δp_2 (item (9)) is increased to 15 inches of water and that the other intercooler limitations, the desired intercooler dimensions including N, and the engine characteristics remain the same. It is desired to find the effect of this increase of Δp_2 on the cooling effectiveness and the characteristics of the intercooler.

(35) $\sigma_2 \Delta p_2 = 15$ inches of water

(36) Figure 16(a) gives the percentage to be added algebraically to the cooling effectiveness given by figure 14(c) for $\sigma_2 \Delta p_2 = 10$ inches of water. In figure 16(a) for $l/d = 40$, $\sigma_1 \Delta p_1 = 6$ inches of water, and $\sigma_2 \Delta p_2 = 15$ inches of water, the correction is 2 percent. In case I the effectiveness was 75 percent, so that the new effectiveness is 75 + 2 = 77 percent.

(37) From figure 16(b) the power required to force the charge through the intercooler increases to 3.21 horsepower.

(38) Ndl/M_2 and the tube weight remain the same.

(39) M_1/M_2 remains the same.

(40) From figure 11(a) for $\sigma_2\Delta p_2 = 15$ inches of water and item (38), $\frac{md}{s} = 215$.

(41) From item (40) and the constant values of m and d the tube spacing s = 0.00582 inch.

(42) Because of the closer tube spacing, the entrance-exit loss of the cooling air decreases slightly, so that M_1 actually increases. The changes in M_1 over the range of $\sigma_2\Delta p_2$ covered in figure 16 are negligible, however, so that M_1 and thus the power required to force the cooling air through the intercooler remains substantially constant.

Case IV (sea level)

For the purpose of illustrating the use of figures 8, 9, and 10 and of checking the results obtained from the intercooler design charts (figs. 11, 14, and 15), let it be desired to check the cooling effectiveness η and the pressure drops Δp_1 and Δp_2 of the intercooler having the same physical dimensions and the same rate of cooling air and charge flow as in case I.

From case I the intercooler physical dimensions are:

Item (4), l = 5/6 foot

Item (5), d = 1/48 foot

Item (6), m = 5 banks of tubes

Item (7), t = 0.0005 foot

Item (18), N = 2250 tubes

Item (19), s = 0.000592 foot

Also the charge and cooling-air mass flow are

Item (13), $M_2 = 1.736$ pounds per second

Item (16), $M_1 = 8.0$ pounds per second

(43) From items (5), (16), and (18)

$$\frac{4M_1}{\pi d^2 N} = \frac{4 \times 8.0}{\pi \times \left(\frac{1}{48}\right)^2 \times 2250} = 10.4$$

(44) If figure 12 and items (5), (6), (7), and (19) are used

$$f = 0.76$$

(45) If figure 10(a) and items (43), (44), and an average β of 0.06 are used

$$\sigma_1 (\Delta p_{en} + \Delta p_{ex}) = 1.00 \text{ inch of water}$$

(46) From items (5), (16), and (18)

$$\frac{Nd}{M_1} = 6$$

(47) Then from figure (9) and items (4), (5), and (46)

$$\left(\frac{l}{d}\right)_{eq} = 40 \times 0.90 = 36$$

(48) From figure 10(b) and items (43), (47), and an average $\beta = 0.06$

$$\sigma_1 \Delta p_{fr} = 4.70 \text{ inches of water}$$

(49) From figure 10(c) and item (43) and an average β of 0.06

$$\sigma_1 \Delta p_H = 0.50 \text{ inch of water}$$

(50) The total cooling-air pressure drop obtained by adding items (45), (48), and (49) is:

$\sigma_1 \Delta p_1 = 6.2$ inches of water, as compared with 6.0 inches water used in case I.

(51) The value of Δp_2 will, of course, check because it is taken from the same graph (fig. 11) for the same values of $Nd l/M_2$ and md/s.

(52) From items (13) and (16)

$$\frac{M_1}{M_2} = 4.6$$

(53) With the use of figure 9 and items (5), (6), (19), and (46)

$$\left(\frac{md}{s}\right)_{eq} = 146$$

(54) If items (52) and (53) are used and interpolation is is made between figures 8(a) and 8(b) for $(l/d)_{eq} = 36$ (item (41)), then, $\eta = 75.7$ percent as compared with $\eta = 75$ percent in case I.

Annular Intercooler

A typical annular intercooler set-up is shown in figure 17. The intercooler forms a toroid around the supercharger directly behind the engine cylinders. The engine charge flows radially outward from the supercharger through a suitable number of passages to the annular chamber, from where it flows across a small number of closely spaced tube banks. The charge then flows from the annular chamber into a suitable number of ducts leading to the cylinder intakes. In flight, one portion of the cooling air entering the cowling is deflected by baffles around the cylinder fins for engine cooling and another portion is deflected into the openings between the cylinders. This cooling air flows into the annular opening and through the tubes where it picks up the heat of the charge. It is then discharged into the compartment behind the engine, from where it flows through the cowling exit, together with the engine cooling air.

A comparison of figures 14(c) and 15(c) shows that, for given conditions of pressure drop and mass flow, the cooling effectiveness is increased by reducing the number of tube banks. An example of such a comparison is given in the following table, to be used with figures 14(c) and 15(c):

Number of banks	5	30
$\sigma_1 \Delta p_1$, inches of water	6	6
$\sigma_2 \Delta p_2$, inches of water	10	10
M_1/M_2	4	4
l/d	40	40
Ndl/M_2	20.3	22.5
η, percent	71	66

If d, l, and M_2 remain constant while the number of tube banks is being changed from 5 to 30, the number of tubes and hence the tube weight must be increased in the ratio of 22.5/20.3, as the table shows. At the same time, the cooling effectiveness is reduced from 71 to 66 percent, as shown. Thus, it is seen that the intercooler of the annular type, which in most cases would require a comparatively small number of banks, is slightly advantageous over types having a large number of banks because of the increased cooling effectiveness as well as the reduced tube weight. Also, the annular intercooler receives its cooling air from within the cowling and requires no external air scoop, which would require an additional expenditure of power because of its drag.

CONCLUDING REMARKS

The test results of the test unit closely checked the performance of cross-flow tubular intercoolers calculated from the equations derived from heat-transfer theory. The relative effects on intercooler performance of various intercooler dimensions are easily determined from the design charts presented. The design of an intercooler, which heretofore involved the use of considerable test data and was a trial-and-error process, is greatly simplified by a proper correlation of the variables involved in the present analysis. The resulting design charts show that the power required in forcing the cooling air and the charge through a well-designed intercooler is small, so that this power is of secondary importance to the weight, the size, and the ruggedness of construction of the intercooler. Annual intercoolers with a small number of tube banks are slightly more efficient for a given power expenditure than intercoolers with a large number of banks.

The performance of a cross-flow tubular intercooler is practically independent of the fuel-air ratio of the engine charge at a charge outlet temperature of 140° F.

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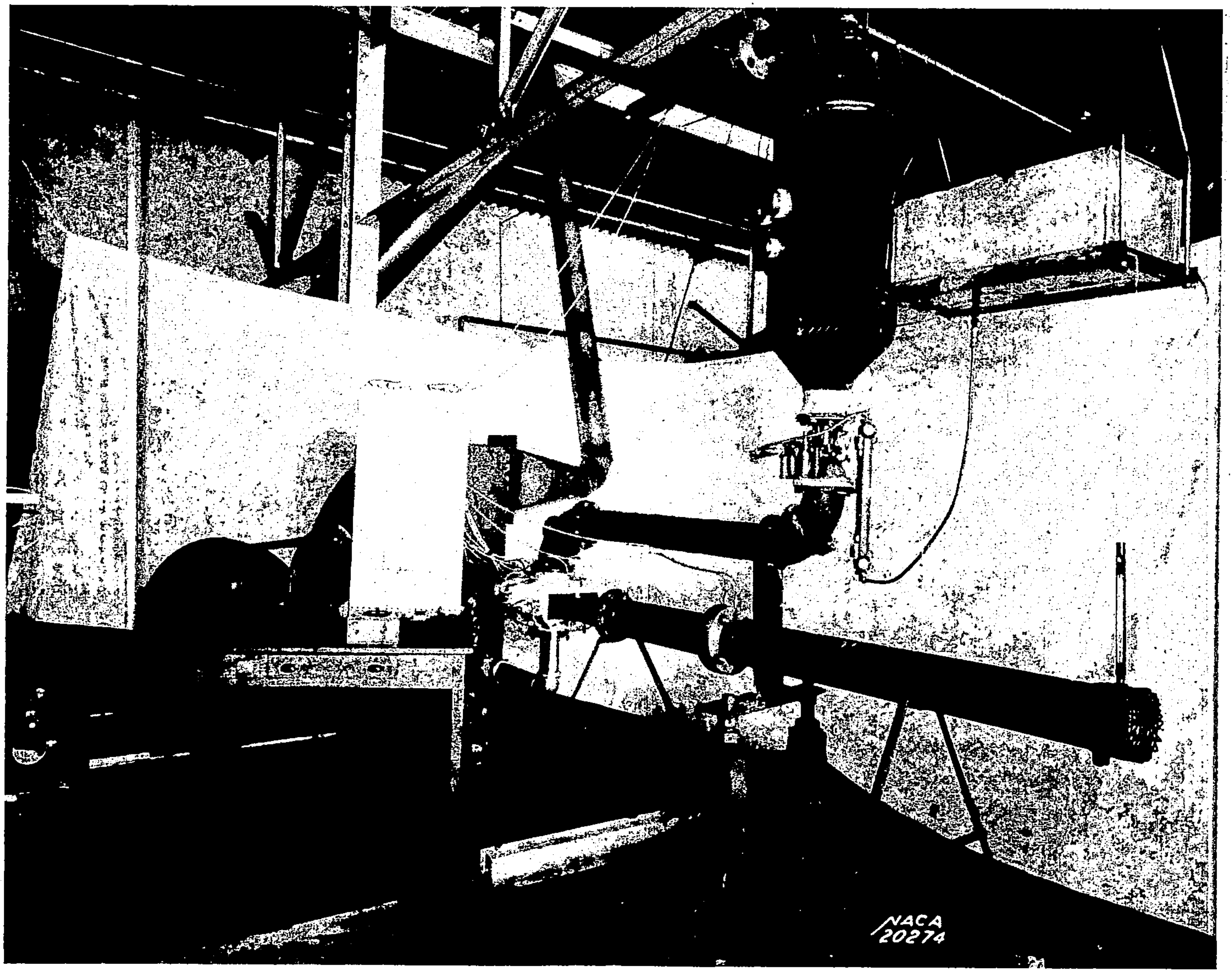


Figure 1.- Test assembly.

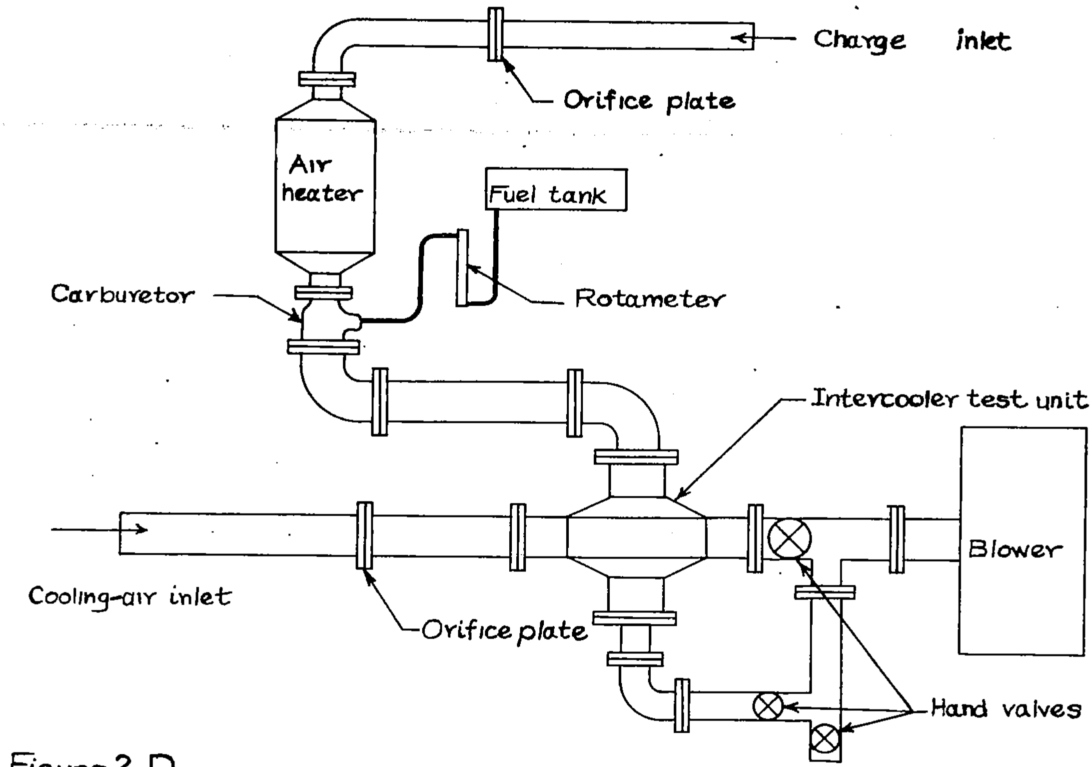
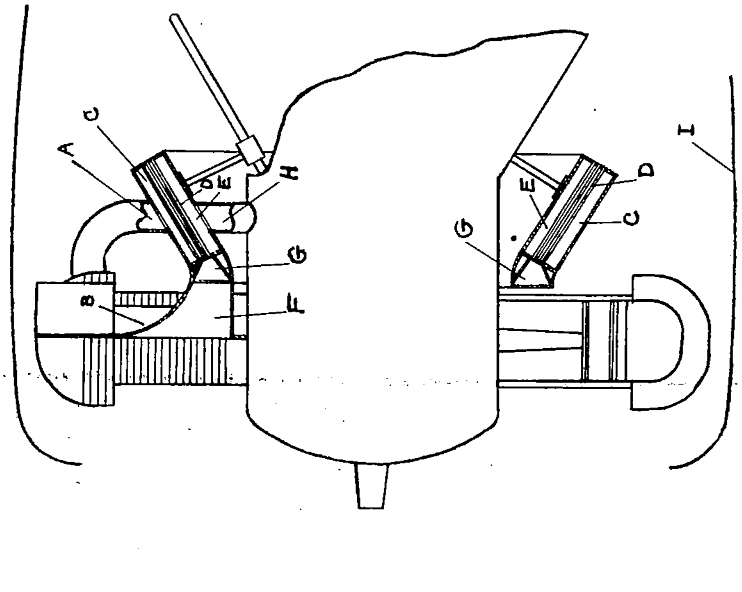


FIGURE 2.-DIAGRAM OF TEST EQUIPMENT.



A, PASSAGE TO ENGINE CYLINDERS
B, ENGINE COOLING-AIR BAFFLES
C, ANNULAR CHARGE EXIT CHAMBER
D, INTERCOOLER TUBES
E, ANNULAR CHARGE ANTECHAMBER
F, COOLING-AIR ENTRANCE
G, ANNULAR COOLING-AIR ANTECHAMBER
H, PASSAGE FROM SUPERCHARGER
I, ENGINE COWLING

FIGURE 17.-TYPICAL ANNULAR INTERCOOLER ASSEMBLY.

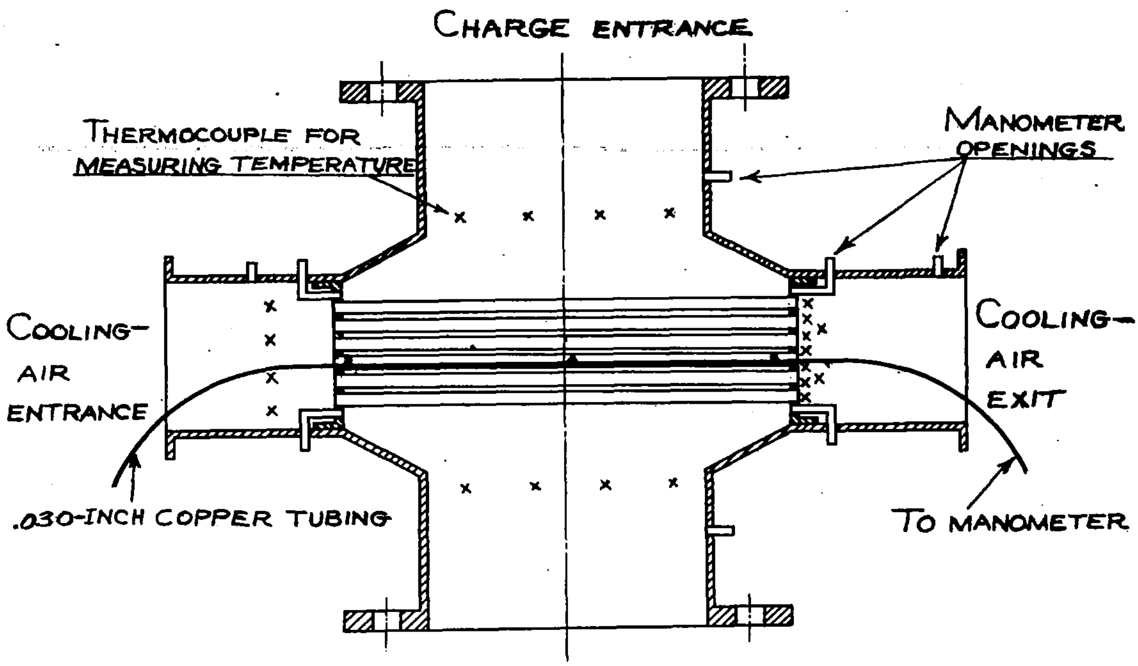
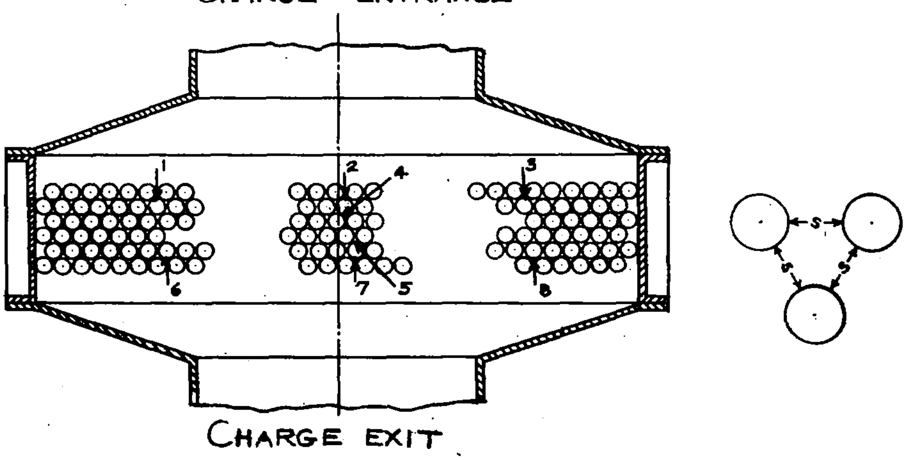


FIGURE 3.- SIDE SECTION OF INTERCOOLER TEST UNIT.



(a) FRONT SECTION OF INTERCOOLER TEST UNIT.

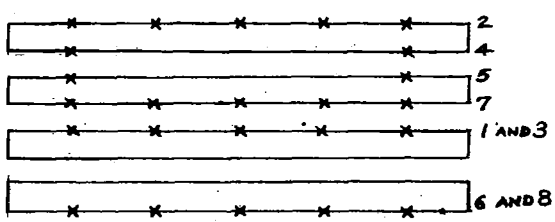
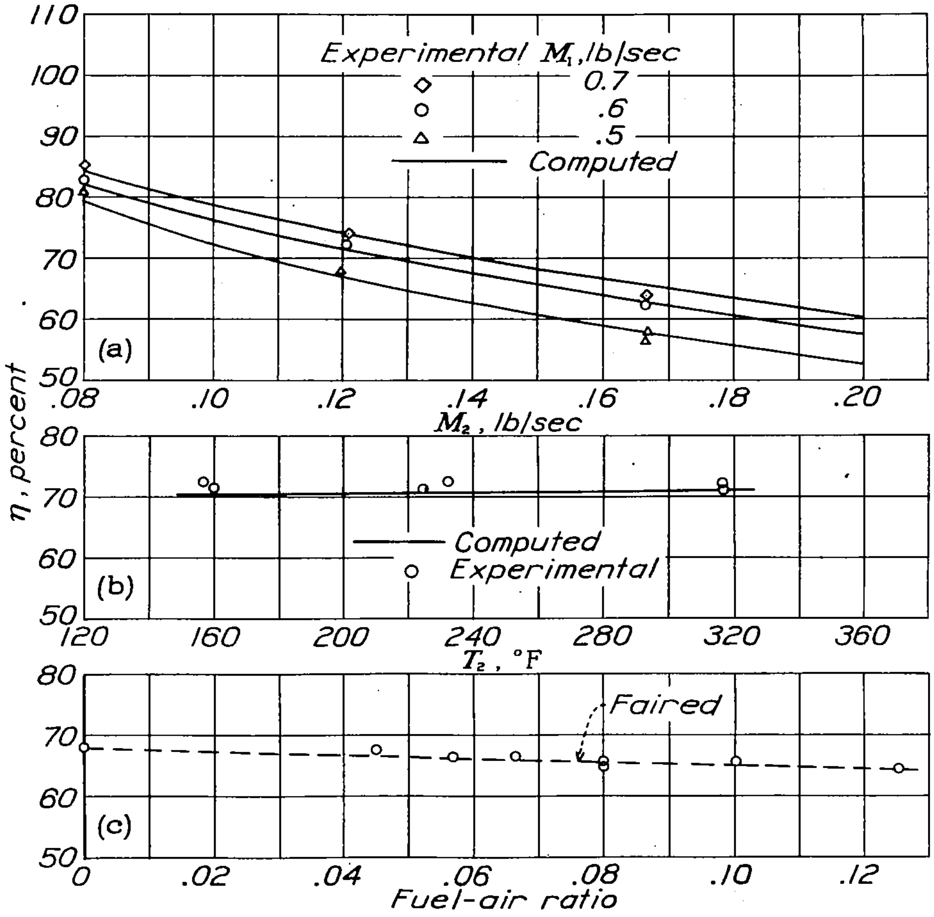


FIGURE. 4.- INTERCOOLER TEST UNIT.

(b) THERMOCOUPLE LOCATIONS ON HEAT-TRANSFER TUBES.



(a) With mass flow of cooling air and charge. Fuel-air ratio, 0 .
(b) With charge inlet temperature. Fuel-air ratio, 0 ; M_1 ,0.60 lb per sec; M_2 , 0.12 lb per sec.
(c) With fuel-air ratio of the charge. M_1 , 0.60 lb per sec; M_2 , 0.12 lb per sec.

Figure 5. - Experimental and theoretical variation in cooling effectiveness of intercooler test unit.

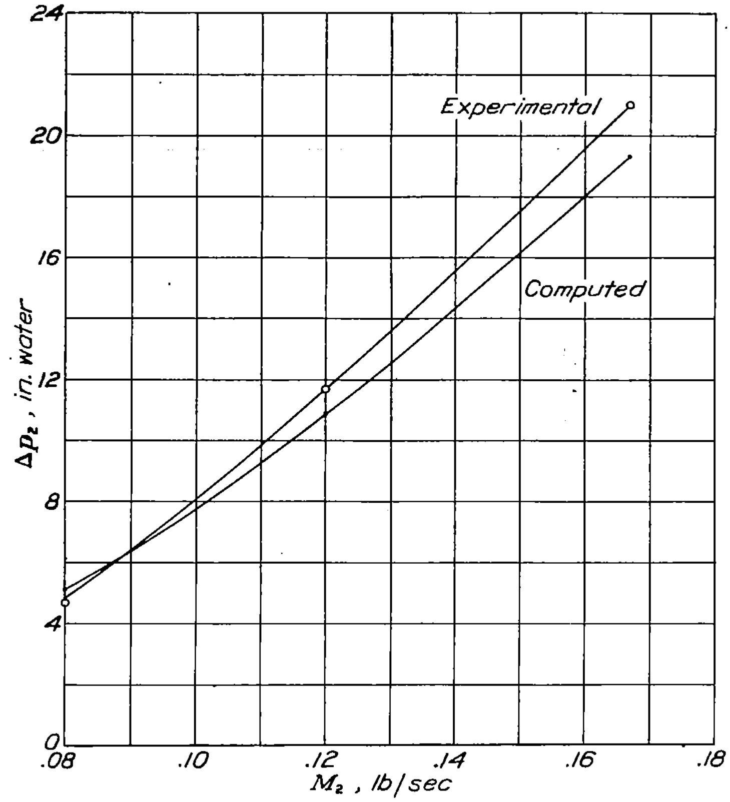


Figure 6.- Experimental and computed variation in pressure drop of charge across the tube banks of the intercooler test unit with mass flow of charge. Fuel-air ratio, 0

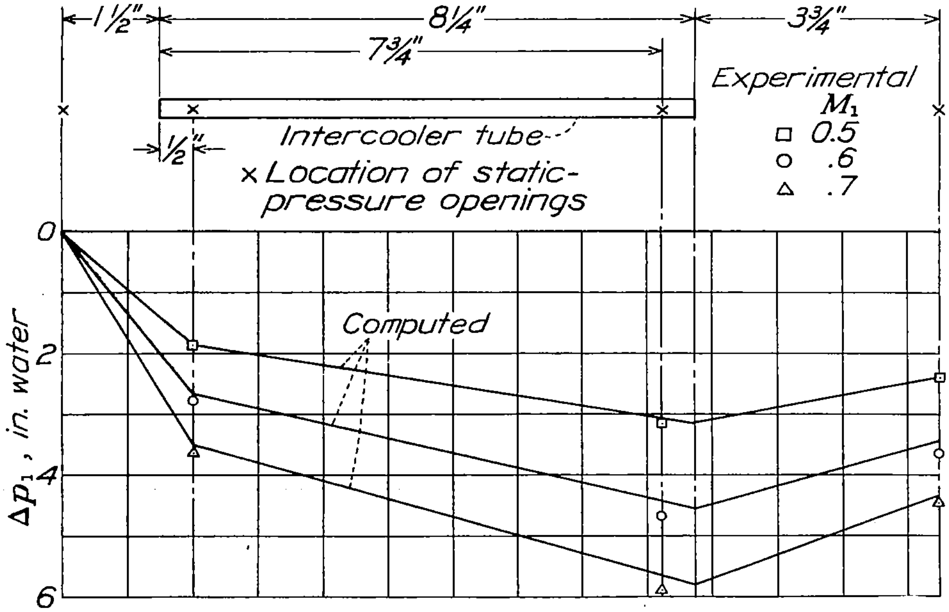
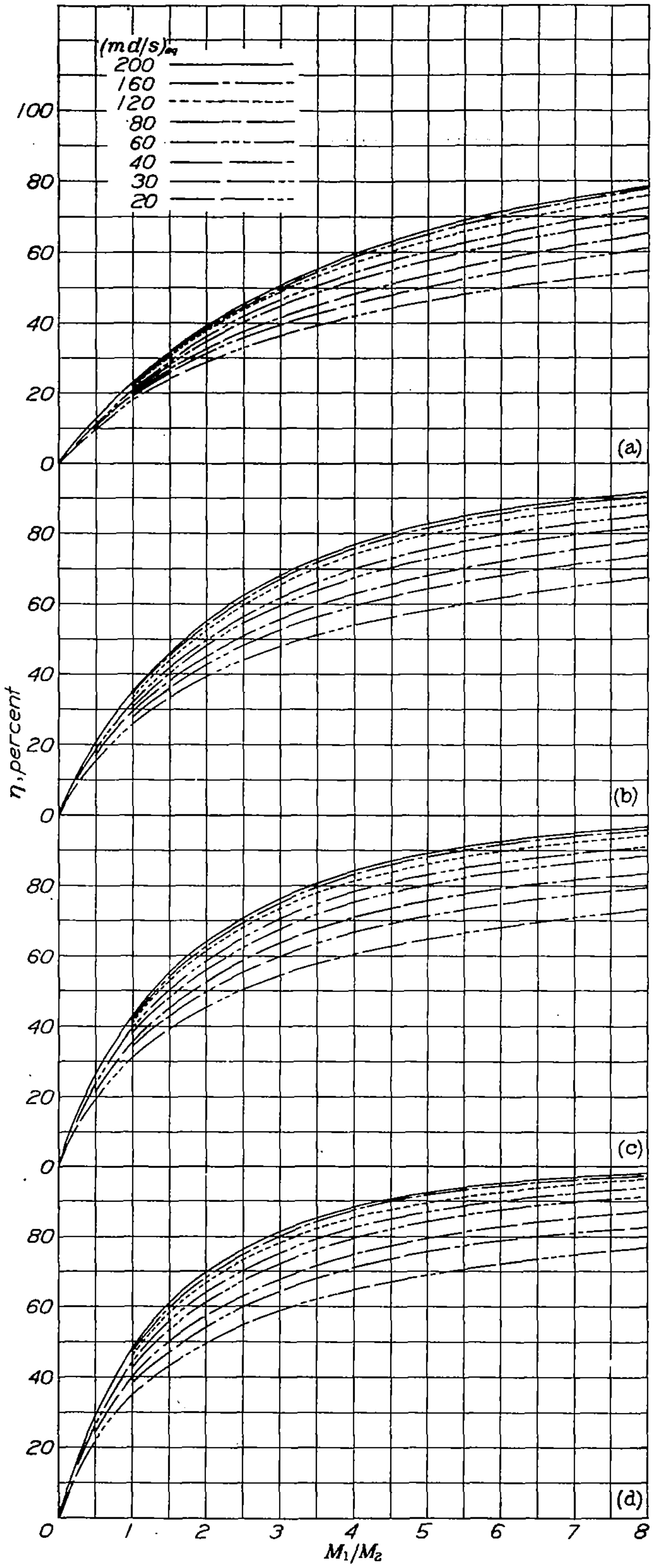


Figure 7.- Schematic diagram showing agreement between experimental and computed pressure drops of the cooling air when it flows through the intercooler test unit.



(a) Equivalent tube length-diameter ratio, $(l/d)_{eq}$, 20.

(b) Equivalent tube length-diameter ratio, $(l/d)_{eq}$, 40

(c) Equivalent tube length-diameter ratio, $(l/d)_{eq}$, 60.

(a) Equivalent tube length-diameter ratio, $(l/d)_{eq}$, 80

Figure 8. - Effect of intercooler dimensions on cooling effectiveness.

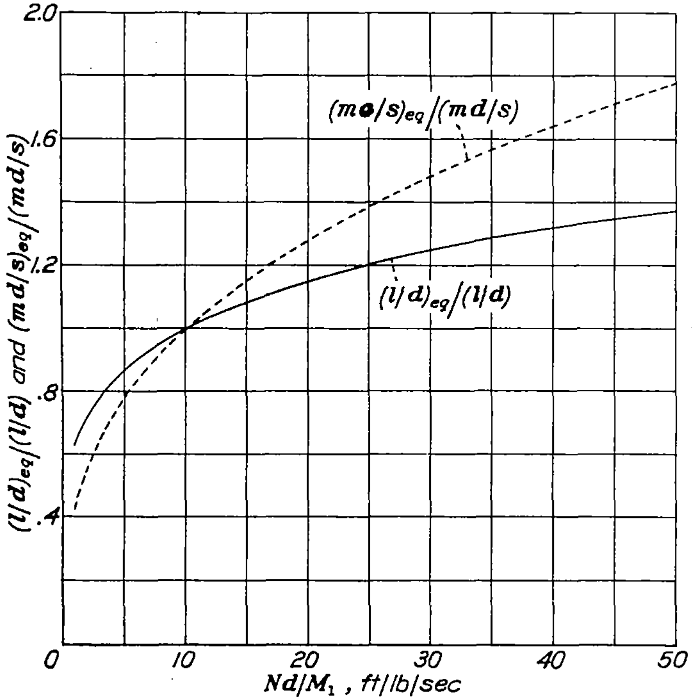
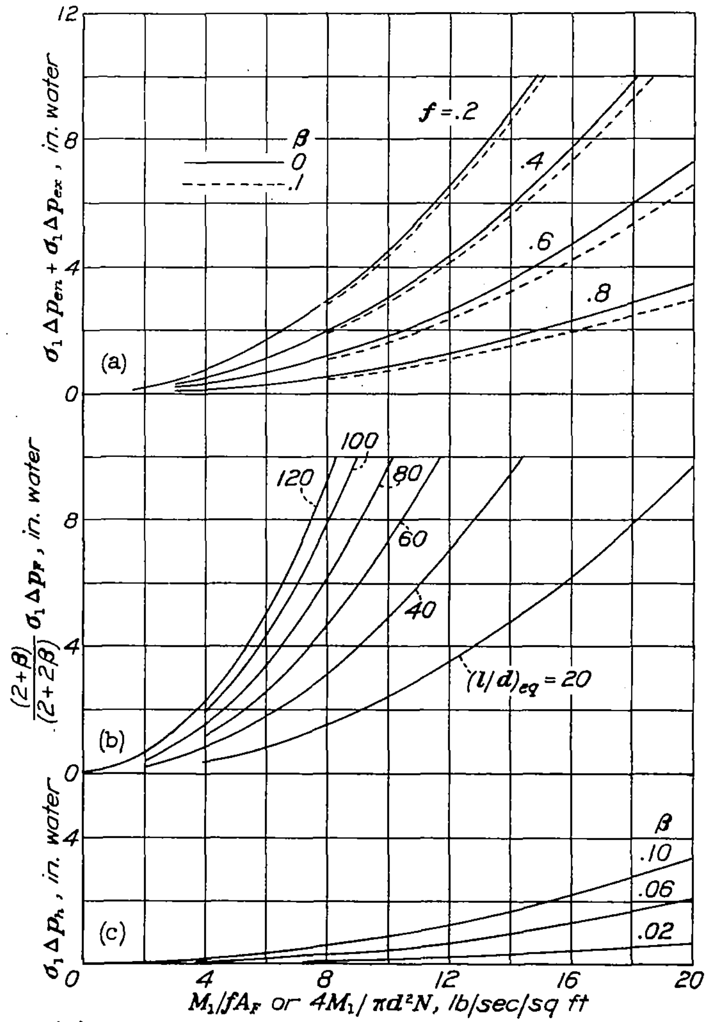
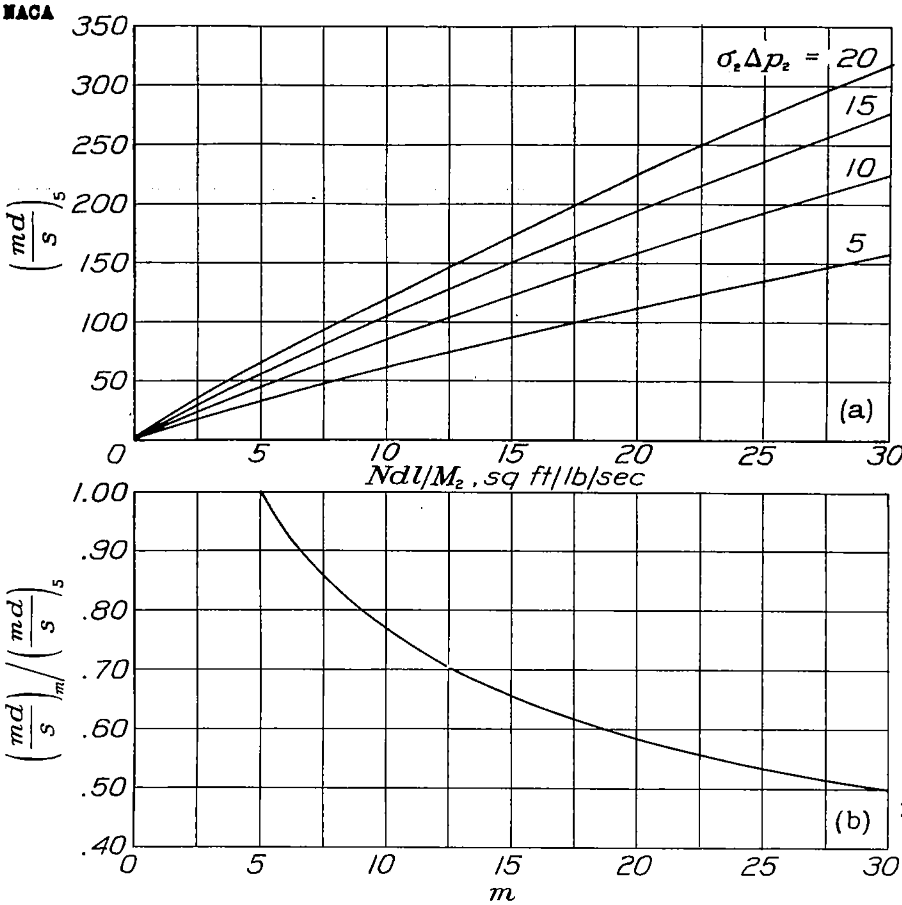


Figure 9.- Correction factors for (l/d) and (md/s).



(a) Entrance and exit loss.
(b) Friction loss in the tubes.
(c) Heating loss in the tubes. $\beta = T_1/T_0+460$.

Figure 10.- Cooling-air pressure drop.



(a) Relation between the charge pressure drop and (Ndl/M_2) and (md/s) for m = 5.

(b) Correction to be applied to $(md/s)_5$ for variation in the number of banks.

Figure 11.- Effect of charge pressure drop and (Ndl/M_2) on (md/s).

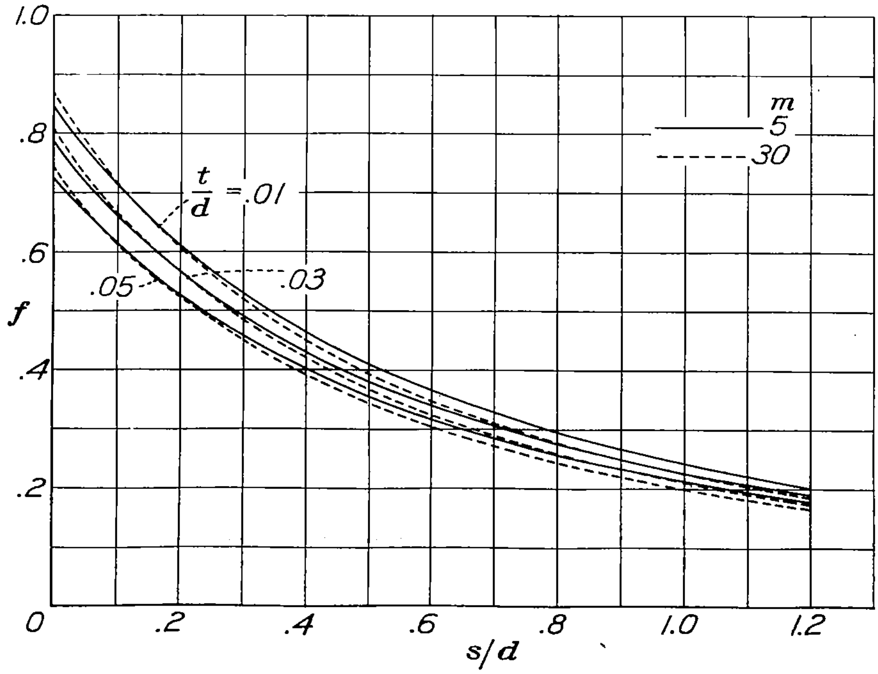


Figure 12.- Effect of intercooler dimensions on area ratio f.

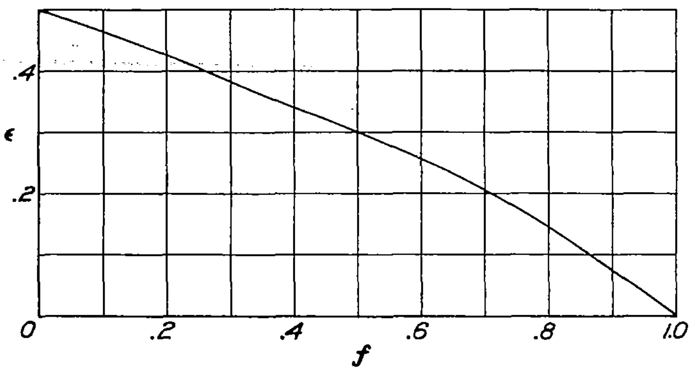
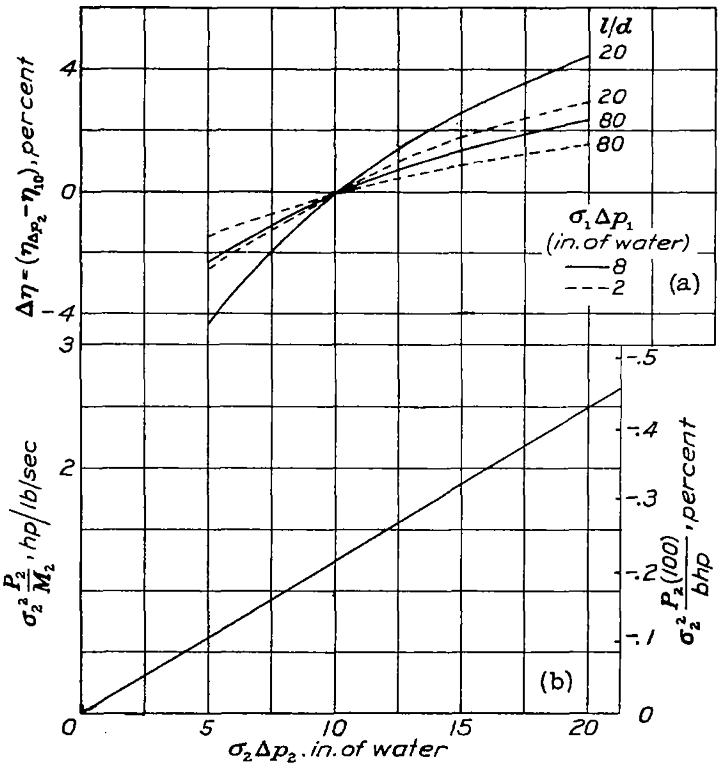


Figure 13.- Effect of area ratio f on contractional-loss coefficient ε.



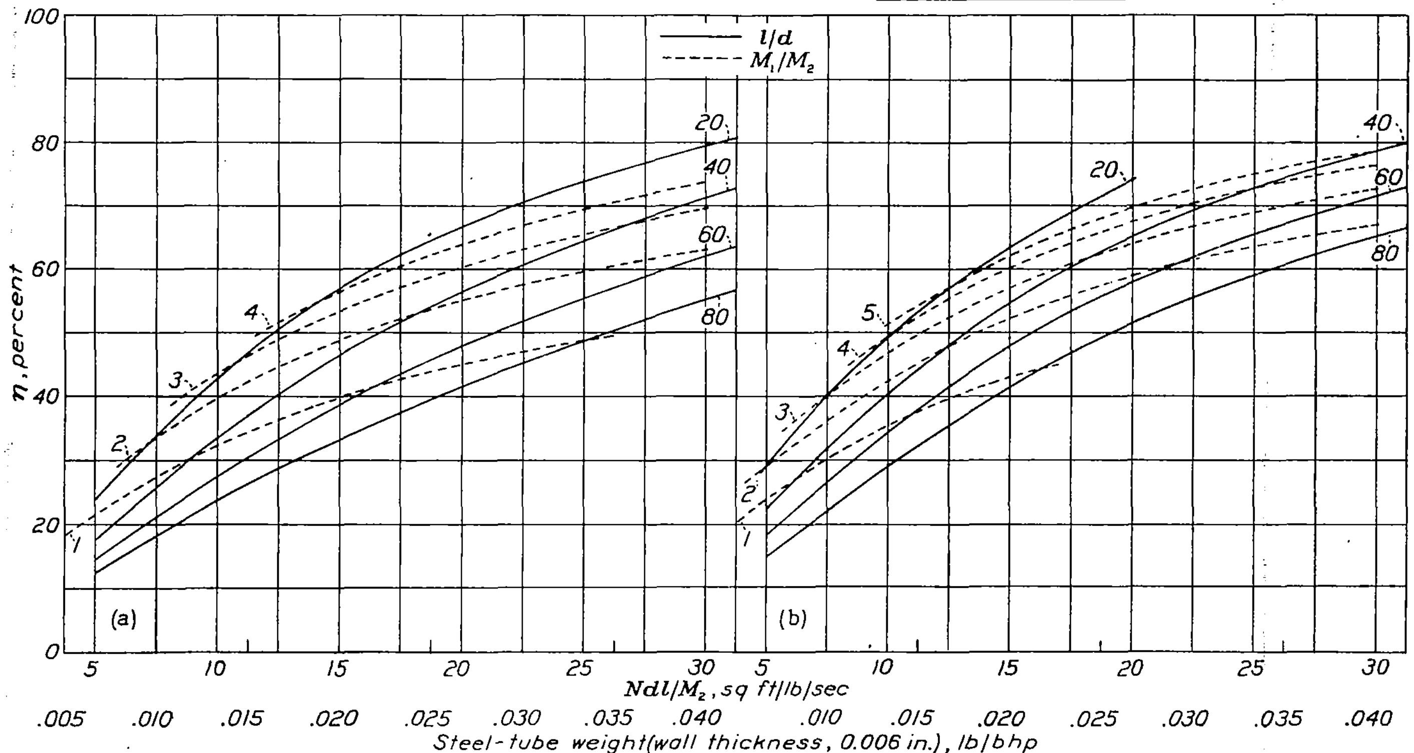
(a) Cooling-effectiveness correction for variations in cooling-air and charge pressure drops and l/d.

(b) Relation between the power required by the supercharger to force the charge across the intercooler and the charge pressure drop.

Figure 16.- Relation between charge pressure drop and intercooler performance.

$\frac{M_1}{M_2}$	$\frac{\sigma_1^2 P_1}{M_2}$ (hp/lb/sec)	$\frac{\sigma_1^2 P_1(100)}{\text{bhp}}$ (percent)
1	0.25	0.044
2	.51	.088
3	.76	.132
4	1.02	.176
	$\frac{\sigma_2^2 P_2}{M_2}$	$\frac{\sigma_2^2 P_2(100)}{\text{bhp}}$
	1.24	0.214

$\frac{M_1}{M_2}$	$\frac{\sigma_1^2 P_1}{M_2}$ (hp/lb/sec)	$\frac{\sigma_1^2 P_1(100)}{\text{bhp}}$ (percent)
1	0.51	0.088
2	1.02	.176
3	1.52	.264
4	2.03	.352
5	2.54	.440
	$\frac{\sigma_2^2 P_2}{M_2}$	$\frac{\sigma_2^2 P_2(100)}{\text{bhp}}$
	1.24	0.214

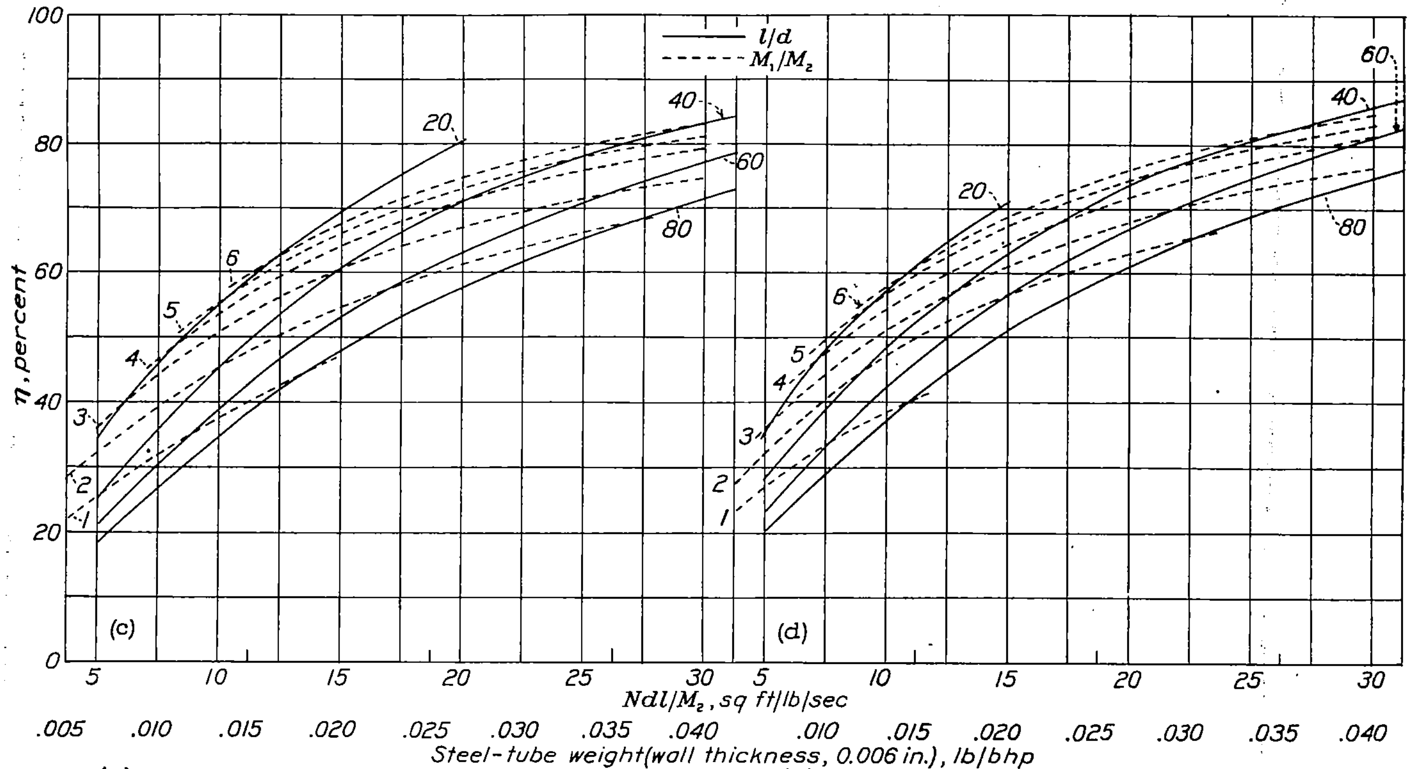


(a) $\sigma_1 \Delta p_1$, 2 inches of water. (b) $\sigma_1 \Delta p_1$, 4 inches of water.

Figure 14 a to d. Intercooler design chart. m, five banks; $\sigma_2 \Delta p_2$, 10 inches of water.

$\frac{M_1}{M_2}$	$\frac{\sigma_1{}^2 p_1}{M_2}$ (hp/lb/sec)	$\frac{\sigma_1{}^2 p_1 (100)}{\text{bhp}}$ (percent)
1	0.76	0.132
2	1.52	.264
3	2.29	.397
4	3.05	.529
5	3.81	.661
6	4.57	.794
	$\frac{\sigma_2{}^2 p_2}{M_2}$	$\frac{\sigma_2{}^2 p_2 (100)}{\text{bhp}}$
	1.24	0.214

$\frac{M_1}{M_2}$	$\frac{\sigma_1{}^2 p_1}{M_2}$ (hp/lb/sec	$\frac{\sigma_1{}^2 p_1 (100)}{\text{bhp}}$ (percent)
1	1.02	0.176
2	2.03	.352
3	3.04	.529
4	4.06	.705
5	5.07	.881
6	6.09	1.057
	$\frac{\sigma_2{}^2 p_2}{M_2}$	$\frac{\sigma_2{}^2 p_2 (100)}{\text{bhp}}$
	1.24	0.214

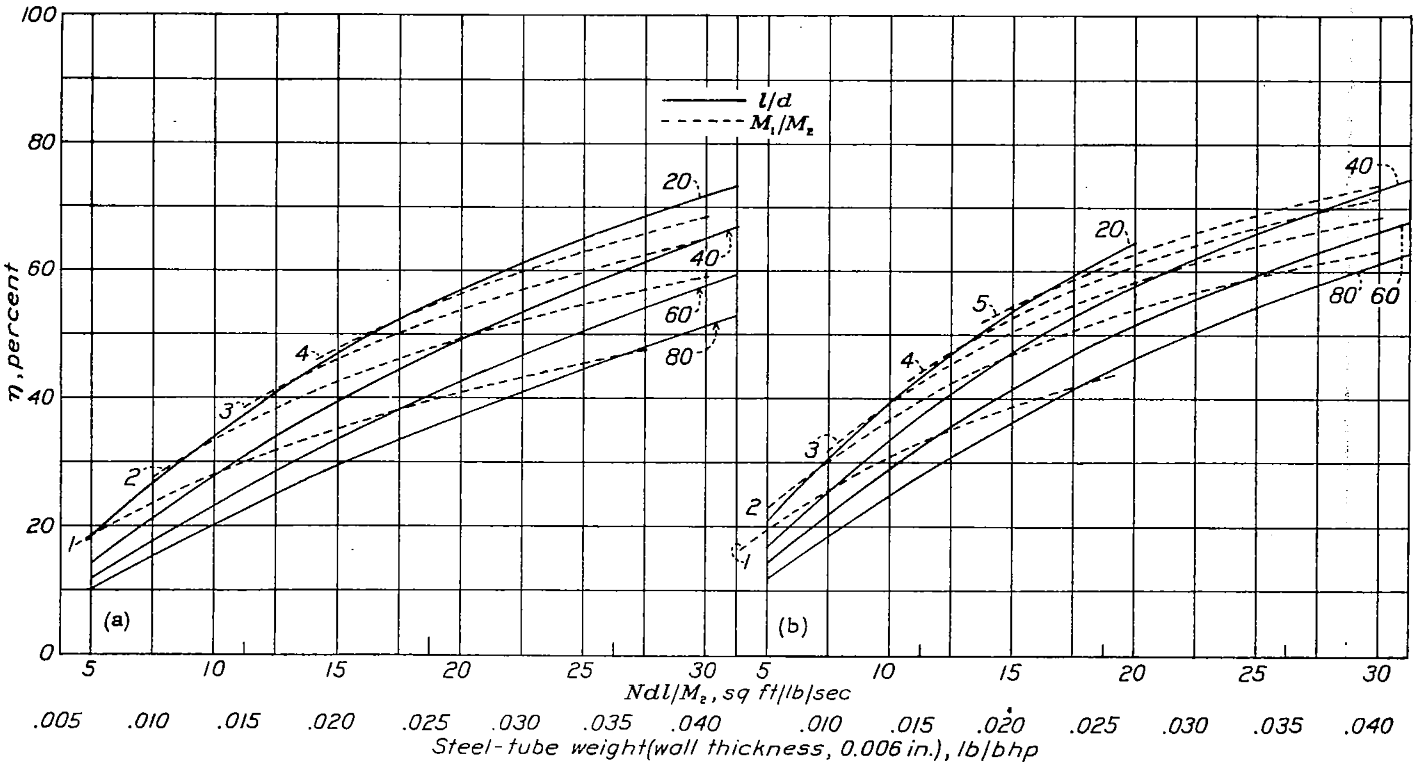


(c) $\sigma_1 \Delta p_1$, 6 inches of water. (d) $\sigma_1 \Delta p_1$, 8 inches of water.

Figure 14. Concluded.

$\frac{M_1}{M_2}$	$\frac{\sigma_1{}^2 P_1}{M_2}$ (hp/lb/sec)	$\frac{\sigma_1{}^2 P_1(100)}{\text{bhp}}$ (percent)
1	0.25	0.044
2	.51	.088
3	.76	.132
4	1.02	.176
	$\frac{\sigma_2{}^2 P_2}{M_2}$	$\frac{\sigma_2{}^2 P_2(100)}{\text{bhp}}$
	1.24	0.214

$\frac{M_1}{M_2}$	$\frac{\sigma_1{}^2 P_1}{M_2}$ (hp/lb/sec)	$\frac{\sigma_1{}^2 P_1(100)}{\text{bhp}}$ (percent)
1	0.51	0.088
2	1.02	.176
3	1.52	.264
4	2.03	.352
5	2.54	.440
	$\frac{\sigma_2{}^2 P_2}{M_2}$	$\frac{\sigma_2{}^2 P_2(100)}{\text{bhp}}$
	1.24	0.214

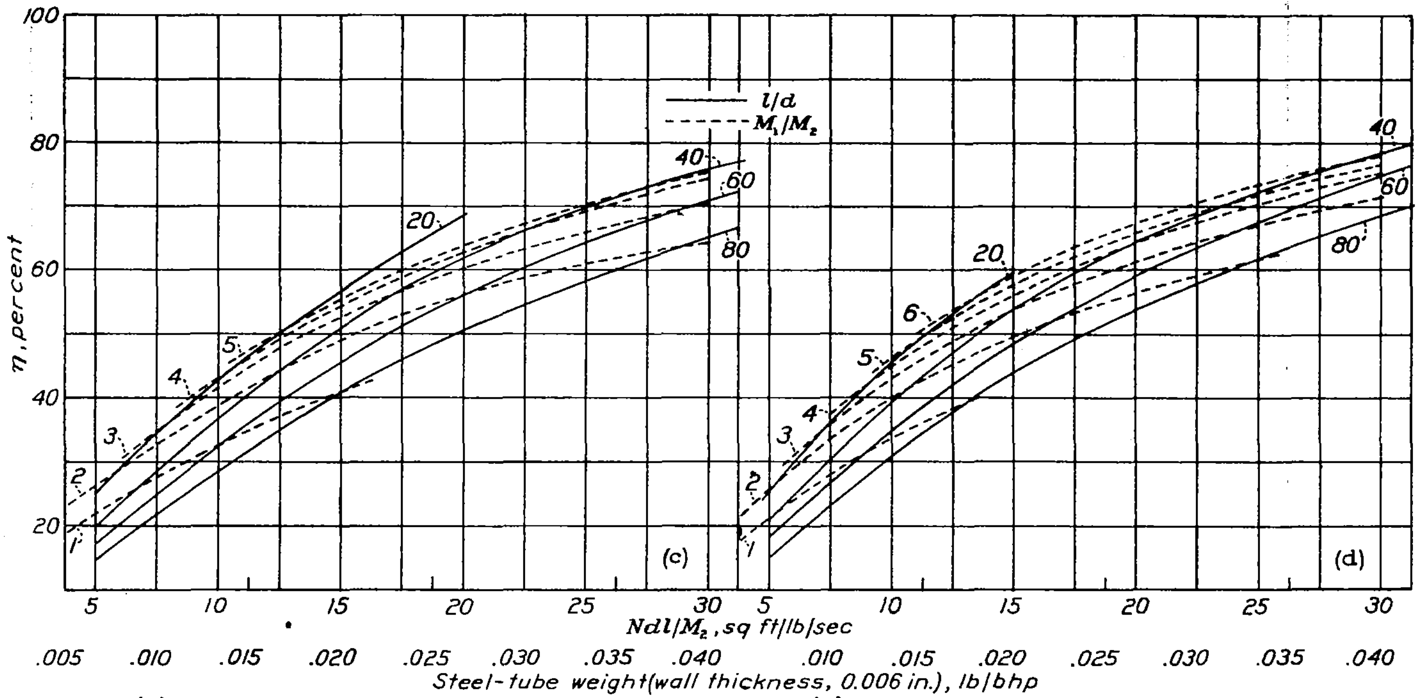


(a) $\sigma_1\Delta p_1$, 2 inches of water. (b) $\sigma_1\Delta p_1$, 4 inches of water.

Figure 15 a to d.- Intercooler design chart. m , 30 banks; $\sigma_2\Delta p_2$, 10 inches of water.

$\frac{M_1}{M_2}$	$\frac{\sigma_1{}^2 P_1}{M_2}$ (hp/lb/sec)	$\frac{\sigma_1{}^2 P_1(100)}{\text{bhp}}$ (percent)
1	0.76	0.132
2	1.52	.264
3	2.29	.397
4	3.05	.529
5	3.81	.661
	$\frac{\sigma_2{}^2 P_2}{M_2}$	$\frac{\sigma_2{}^2 P_2(100)}{\text{bhp}}$
	1.24	0.214

$\frac{M_1}{M_2}$	$\frac{\sigma_1{}^2 P_1}{M_2}$ (hp/lb/sec)	$\frac{\sigma_1{}^2 P_1(100)}{\text{bhp}}$ (percent)
1	1.02	0.176
2	2.03	.352
3	3.04	.529
4	4.06	.705
5	5.07	.881
6	6.09	1.057
	$\frac{\sigma_2{}^2 P_2}{M_2}$	$\frac{\sigma_2{}^2 P_2(100)}{\text{bhp}}$
	1.24	0.214



(c) $\sigma_1 \Delta p_1$, 6 inches of water. (d) $\sigma_1 \Delta p_1$, 8 inches of water.

Figure 15.- Concluded.